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Effect of operating conditions on performance of domestic heating systems with heat pumps and fuel cell micro-cogeneration.

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Abstract

The relative performances of six Air Source Heat Pumps (ASHP) and a Solid Oxide Fuel Cell micro-Combined Heat and Power (SOFC-mCHP) unit are compared using a modelling approach. The emphasis is in indicating the effect of a wide range of operating conditions and methodologies, rather than detailed analysis of the performance of the units under limited specific circumstances. The effect of control methodologies is the primary focus but other variables such as the climate and the specification of the buildings to which heat is supplied are considered. Several significant findings emerge. Firstly, a reduction in heating demands due to warmer will reduce the impacts of both heating systems. In the case of ASHPs, lower heat demands improve performance. In the case of SOFC-mCHP systems they reduce the need for auxiliary heating. A wide range of performances may be achieved by ASHPs, even supplying heat to the same building; the way in which ASHP units are controlled has the potential to reduce their impacts by more than a third. The greatest savings achieved by the SOFC-mCHP unit occur when it is run continuously at full output, despite the consequent dumping of excess heat. Although the auxiliary heaters used with them inevitably reduce their overall

benefit, they are still capable of significant savings. It is currently possible for the units to offset more emissions than they create.

keywords: air source heat pump, solid oxide fuel cell micro-combined heat and power, domestic heating, efficiency, performance

Abbreviations:

ASHP Air Source Heat Pump

COP Coefficient of Performance

mCHP micro Combined Heat and Power

SOFC Solid Oxide Fuel Cell

1. Introduction

To achieve ambitious reductions in greenhouse gas emissions, nations with temperate climates will need to decarbonise the way in which domestic space heating is delivered [1]. Air source heat pumps (ASHPs) and solid-oxide fuel cell micro-combined heat and power (SOFC-mCHP) units have been suggested as two technologies with the potential to contribute towards achieving this [2]. Although many studies have investigated the performance of units in detail, this study considers their relative performance under a

wider set of configurations and conditions in order to investigate the effect that these factors may have.

Extensive testing of individual units has been conducted (e.g. [3]) but it is important to consider the effect of a wider range of operational conditions. Field trials (e.g. [4–6]) provide valuable data which has been analysed to suggest potential areas for improvement (e.g. [7,8]) but are generally limited in the scope of the options which they can consider. To address this, detailed modelling has been used by several researchers to analyse the potential performance of different low-carbon heating technologies in various configurations. These studies typically provide an overview of the relative merits of the technologies. Some focus in detail on the performance of a single technology in a specific context and compare this to the default alternative (e.g. [9]) whilst others compare the performance of different technology options (e.g. [10–14]). Consideration is usually given to the effect of the source of central electricity generation and the specification of the buildings to which heat is supplied. However, whilst different control configurations and climates were used in the studies, indicating that they have an effect, these effects have not yet been explored fully. Although the optimum control of individual units has received attention (e.g. [15]), studies which investigate the effect of the control and configuration of the units in the context of the overall systems in which they operate are surprisingly few. Madani et al. [16] showed the potential for different control techniques to improve heat pump performance but focussed solely upon techniques that take flow temperatures as inputs.

The approach taken in this study was to simulate the relative performance of heating system (i.e. the units and their auxiliary systems) operating under a wide range of operational conditions rather than to focus on detailed simulation of the impacts of the units under specific conditions. The parameters

which were identified as being of interest were the control methodologies used by the heating systems and the climate in which they operate. Different building specifications were additionally simulated, providing a consistent comparison across these parameters. In addition to characterising the effect of these parameters, the potential of appropriate control systems to achieve significant reductions in energy demand and emissions was demonstrated.

2. Method

2.1 Overview of conditions investigated

The effect of a wide range of operating conditions on the performance of the units was investigated by modelling the systems and then simulating them under 267 permutations representing the scenarios and operational parameters detailed below. Performance was considered in terms of efficiency and greenhouse gas emissions (see Section 2.2). The performance of six ASHPs, a SOFC-mCHP unit and a condensing gas boiler were compared (see Section 2.3). The permutations were arranged in groups:

- 147 permutations were formed from six models of air source heat pumps and a condensing gas boiler in three building specifications with seven combinations of control methodology and buffer tank capacity. These options are described in Section 2.4.
- 72 permutations were formed by simulating a SOFC-mCHP unit with different control methodologies and configurations. The first 36 consisted of six buffer sizes and six control methodologies. An additional 36 permutations were formed from two variations on the highest performing methodology analysed earlier with three building specifications and six buffer tank sizes. These are described in Section 2.5.

- Finally, 48 permutations considered the potential effect of climate change on the performance of the units. Simulations of two ASHPs, the SOFC-mCHP unit and a condensing gas boiler were conducted using data for 12 climates. The selected climate data is described in Section 2.6.

The three building specifications were constructed to be representative of a semi-detached house, a semi-detached house with enhanced heat emitters (effectively an underfloor system) and the same house with enhanced heat emitters and reduced heat losses. These are described in Section 2.7.

2.2 Performance metrics

Results are based upon the total annual energy flows. Efficiency calculations for the SOFC-mCHP unit used the gross calorific value of fuel input and their alternating current electrical output (i.e. net of inverter losses). ASHP performance is expressed as a Coefficients of Performance (COP, i.e. the quotient of heat delivered by an ASHP to electrical work required). *Unit* performance metrics were based upon energy flows to and from the individual units. *System* performance metrics were based upon the heat flows to the hot water tank and heat emitter system and the fuel and net electrical inputs to both the units and their auxiliary systems (i.e. including auxiliary heaters and pumps).

Greenhouse gas emissions were also used to assess system performance. An emissions factor of 189gCO₂e/kWh was used for natural gas, based upon its content and transmission efficiencies [17]. Emissions upstream of entry to the national transmission system were not included. An electrical grid carbon emissions factor of 586gCO₂e/kWh was used based upon fixed emissions characteristics for each generation type [18,19] with the mix of generation weighted by heat demand using time-series generation data [20].

It should be noted that operational emissions were used; if the aim of a study were to provide a full comparison between micro-generation systems it would be necessary to complete a full life-cycle assessment of their impacts [21,22]. Results comparing the emissions associated with ASHPs and mCHP units are very sensitive to the carbon emissions factors which are assumed but this is explored in detail elsewhere and is not pursued further in this study [10–14,23].

2.3 Heating system performance

The nominal COPs of the ASHPs are given in Table 1. These figures relate to standardised test conditions [24] but the sources referred to include performance data at between eight and twelve additional sets of conditions for each unit.

Table 1: Nominal ASHP performances

Unit	COP	Reference
ASHP A	4.2	[25]
ASHP B	3.0	[26]
ASHP C	3.6	[27]
ASHP D	3.5	[27]
ASHP E	3.4	[25]
ASHP F	4.4	[25]

An interpolation method was used to determine the performance of the ASHP units. The exergy efficiency of each unit was calculated at each of the standardised test conditions for which data was available [25–27]. The weighted average of the exergy efficiencies at the four test conditions with source and sink temperatures bounding the temperatures in the model was calculated during each time step. This exergy efficiency was then used to calculate the power consumption under those conditions. Some studies [13,28] have successfully applied parametric relationships between the performance of ASHP units and the temperatures they operate between. The method used here takes advantage of the observation that the exergy efficiency of heat

pumps tends to be approximately constant between test conditions [29] in order to improve confidence in the model when the simulated conditions tended towards the more extreme test conditions. The heat which was generated by each heating unit was also constrained by its maximum and minimum heat generation capacity.

The efficiencies of the SOFC-mCHP unit are given in Table 2 for two electrical output levels. Because of the low heat generation capacity of the SOFC-mCHP unit and its slow ramp-rate, its operation was supplemented by an auxiliary gas boiler. The heat from both units fed into a buffer tank.

Table 2: Steady-state unit efficiencies of SOFC-mCHP unit [30]

	Electrical (net)	Thermal
Peak electrical efficiency (1.5kWe)	54%	21%
Peak electrical generation (2.0kWe)	51%	29%

The steady-state electrical and thermal efficiencies of the SOFC-mCHP unit were calculated as a function of the output level by linear interpolation from a set of published performance data [30]. A default maximum ramp-rate of 0.06W/s was assumed based upon known warm-up and cool-down times. The results demonstrate relatively low sensitivity to this assumption (see Section 3.2).

The gas boiler system was modelled with a fixed thermal efficiency of 90%. In reality, the efficiency achieved by such devices is a function of the flow temperature they operate with (and the extent to which condensing operation is therefore achievable) but the fixed efficiency provides a more transparent comparison to the other systems.

2.4 ASHP control methodologies

The 147 permutations of ASHP configurations were formed from six ASHP units, three building specifications and seven combinations of control methodology and buffer tank capacity. These seven combinations were:

- Fixed-temperature control. The controller aimed to maintain the buffer tank at a fixed temperature. For buildings using standard heat emitters, the target buffer tank temperature was 55°C. For buildings using enhanced heat emitters the target buffer tank temperature was 40°C. Buffer tank capacities of 40kg, 80kg and 160kg were used. A thermostat with a $\pm 1^\circ\text{C}$ deadband controlled the flow of heat from the buffer tank to the heat emitters.
- Variable-temperature (“weather-compensated”) control. The controller aimed to maintain the temperature of the buffer tank at a temperature which was determined as a function of the outside air temperature (see Figure 1). Again, buffer tank capacities of 40kg, 80kg and 160kg were used with thermostat flow control.
- Proportional control. The target heat generation was calculated as a function of the difference in temperature between the temperature programme set-point and the air temperature at a point inside the dwelling. Buffer tanks were not used.

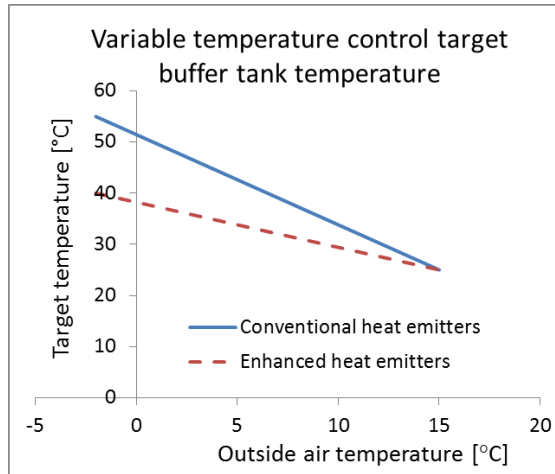


Fig. 1: Temperature control

2.5 SOFC-mCHP control methodologies

The first 36 permutations considered with the SOFC-mCHP unit consisted of six control approaches with six buffer tank sizes. The control methodologies were:

- Full range, heat-led operation. Minimum output was taken to be 320W thermal, 200W electrical.
- Continuous operation at maximum electrical output (2kW electrical, 1kW thermal).
- Continuous operation at maximum electrical efficiency (1.5kW electrical, 530W thermal)
- Constrained to only operate between maximum electrical efficiency and maximum electrical output. Heat-led within this range.
- Full range, heat-led operation but with limit on ramp-rate hypothetically removed.
- Operation between maximum electrical efficiency and maximum electrical output with limit on ramp-rate hypothetically removed.

These were simulated with buffer tank capacities of 40kg, 80kg, 160kg, 320kg, 640kg and 1280kg. In each case, the auxiliary boiler operated to meet any heat demand which

exceeded that which the mCHP unit could deliver by more than 500W. Heat demand was determined by the same variable-temperature control method as that used by the ASHP systems.

The second set of 36 permutations involving the SOFC-mCHP units compared the use of the variable-temperature control method and the fixed-temperature control method for determining the required heat generation. The SOFC-mCHP units were constrained to operate at their peak electrical output and so fluctuations in the heat demand were met by the auxiliary boiler. This was repeated with the six buffer tank capacities listed above and with the three building specifications used with the ASHP simulations.

2.6 Climate data used

The relative effect which changes in climate may have was studied by modelling the operation of the SOFC-mCHP unit, two ASHPs and a gas boiler with 12 different climates.

Four locations across the United Kingdom were selected due to their different characteristics: Cardiff (south west, coastal), London, Leicester (midlands, England) and Glasgow (west Scotland, coastal). The objective was to draw broader observations which will be generally applicable rather than to make predictions about specific locations.

For each of these locations, hourly climate data for three time periods was taken from the work of the Prometheus project [31]. Test reference year data for the period 1960 – 1990 was used alongside that modelled by the project for the 2030s and 2050s (taking median profiles from mid-estimate, “a1b”, emissions scenario).

2.7 Thermal models

Lumped thermal capacitance models representative of the building and heating systems were used (Figure 2). Similar models

have been used in previous work [32] and a description which includes details of all parameters is available [23].

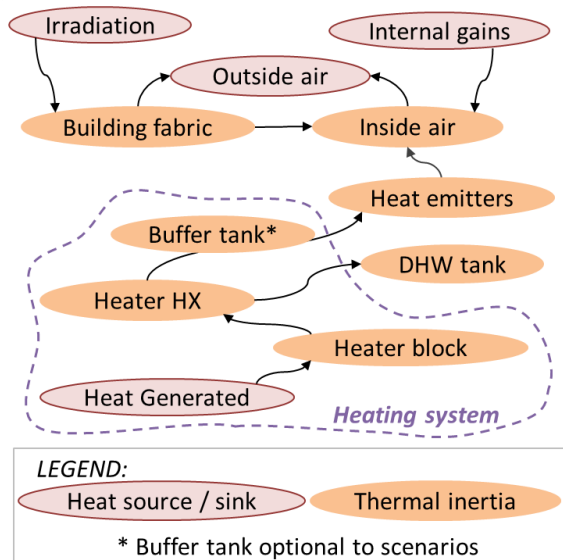


Fig. 2: Thermal model

Testing of heating system models which have been developed with a “two thermal capacity” approach has shown that they are well suited to capturing the dynamics of mCHP operation [11,33]. A similar approach has been successfully used with a non-modulating heat pump [9]. In the present study, the approach is taken primarily so that the relevant flow temperatures can be accurately modelled as these are critical to the performance of heat pumps. Although this approach could enable consideration of the thermal dynamics of the systems, they are less significant than in the other studies mentioned. The heat pumps are capable of modulating their heat generation and have far lower effective thermal inertias than the mCHP units. The dynamics of the SOFC-mCHP unit are dominated by the constraints imposed by its control system rather than the thermal lag of the heat exchangers.

The thermal model of the building was selected to provide a good compromise between accuracy and complexity. Similar models have been shown to be capable of providing adequately accurate profiles and are

appropriate to studies requiring substantial numbers of simulations [34–36]. This approach was well suited to the present study as it did not need to accurately predict the absolute value of heat flows, but rather the dynamism of the thermal transfers in a way which was consistent between simulations. The relative performance of the heating systems under different conditions was of interest, not the heat demand associated with the building (which in any case is highly sensitive to uncontrolled variables such as occupant behaviour). The lower computational overhead which resulted from modelling the building at a similar level of complexity to the heating system facilitated the simulation of a greater range of permutations than that which is typical in studies that employ detailed building models.

Temperature and heat flow profiles of a “standard” semi-detached house [37] have been generated using a detailed model created in ESP-r by Dr. N. Kelly and Dr. J. Hong of ESRU, University of Strathclyde. These data were used to calibrate the various parameters of the simplified model used here. A validation test resulted in a root mean squared temperature difference of less than 0.5°C between the inside air temperature profile of the detailed and simplified models. In the “reduced heat loss” building, the air infiltration rate was halved relative to the calibrated set of parameters.

Heat transfer from the heat emitter system was assumed to follow a buoyancy driven convection model [38]. The nominal effective heat transfer coefficients of the system were defined such that a flow temperature of 50°C for the standard (radiator) system or 35°C for the enhanced (underfloor) system was required in order to match the heat loss from the building (with an inside air temperature of 21°C and an outside air temperature of -1°C).

Internal gains to the simplified model were based upon the “CREST active occupancy and appliance model” [39], assuming “standard”

metabolic rates for standing and reclining [40] and three residents in each house. Hot water was assumed to be drawn from a 70 litre tank in each case with daily demand consistent with that found in empirical studies [41] and distributed according to the active occupancy [39]. If the hot water tank temperature dropped to 40°C, heat was diverted from the space heating system in order to raise the temperature back to around 55°C.

3. Results and Discussion

3.1 ASHP configuration

Figure 3 shows the heat delivered (for space heating and domestic hot water) and the electricity consumed by the 126 permutations of ASHP installations which were simulated. The same results are aggregated by the building specification (top) and then by the ASHP model (bottom). The annual total heat demand is comparable to the 2009 mean for the UK of 16,220kWh [42]. The simulated heat demand for each building specification is consistent, as might be expected; with lower electrical consumption in the buildings with enhanced heat emitters and proportionally lower heat and electrical demands in the buildings with reduced heat losses. For each building specification, however, a wide range of electrical consumptions are observed, with more than twice the consumption in some cases compared to others with the same building specification.

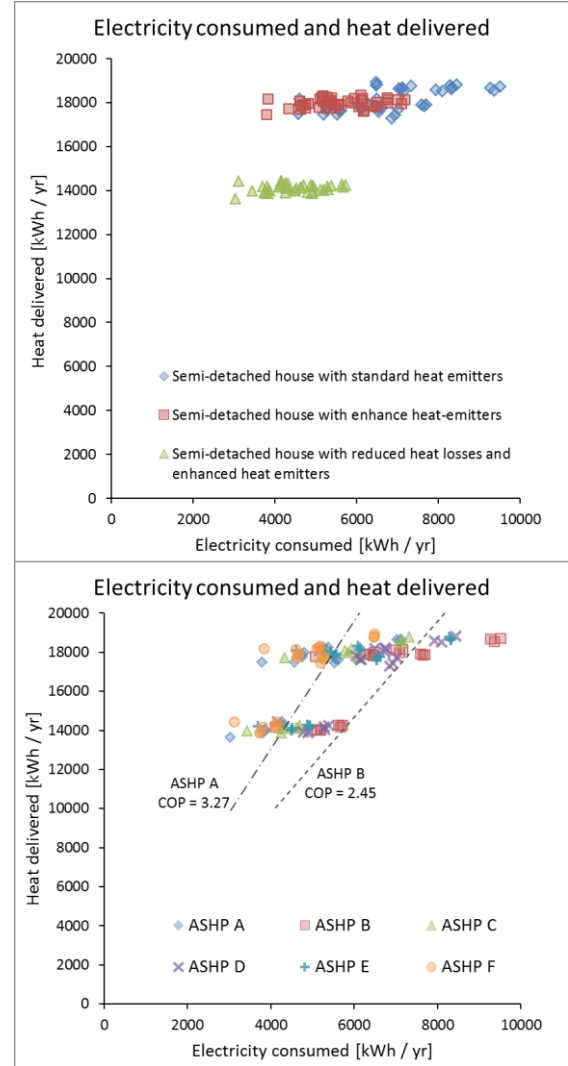


Fig. 3: Heat delivered

Part of this range can be explained by the performance of the ASHP units. The mean COP achieved by ASHP A is just over a third greater than that achieved by ASHP B (a slightly greater difference than implied by their nominal performances, see Table 1). However, the wide spread of electrical consumptions which are observed for each unit are clearly not captured by reference to a single “average” COP characterisation.

To explore this further, Figure 4 illustrates the effect that the different operating conditions have on the performance of the units. Results for the seven combinations of buffer tank size and control methodology are grouped together. Within each of these groups, the three vertical

sub-groups (*i, ii* and *iii*) relate to the results corresponding to each building specification.

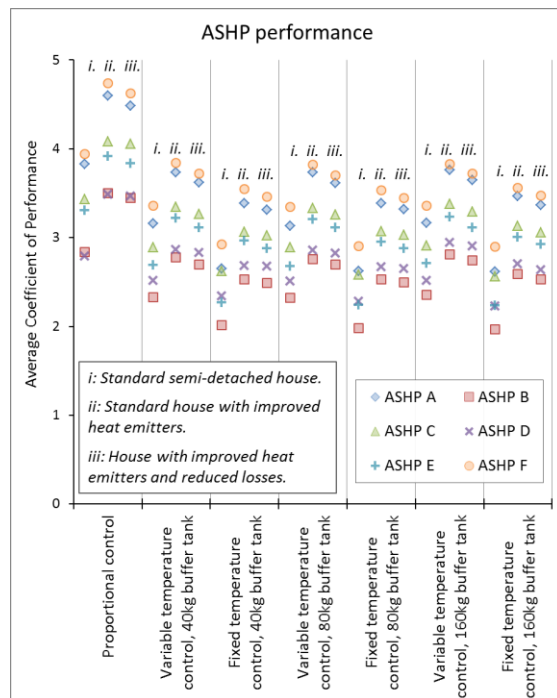


Fig. 4: ASHP performance

Within each sub-group of results, it can be seen that the COP of the highest performing unit (ASHP F) is typically a third higher than the COP of the lowest performing unit (ASHP B). It should be noted that ASHP B is a popular mid-range unit; it does not represent the lowest performing units which are commercially available (and for which reliable test data is typically unavailable). As might be expected therefore, the results of the lower performing units simulated here are consistent with the higher range of performances achieved in field trials conducted by the Energy Saving Trust [4].

Changing the control methodology can improve the COP of the ASHP system by up to 45%. A performance penalty of around 15% to 20% is associated with the use of fixed-temperature control instead of variable-temperature control, consistent with other observations [8,9]. Significantly, an additional improvement of comparable size (15% to 20%) can be achieved by the use of proportional control in the place of variable-

temperature control. The improvement is due to the lower flow temperatures which result from three factors. Firstly, as the use of the outside air temperature to calculate the necessary flow temperature is an approximation, the highest flow temperature is sometimes higher than it needs to be. Secondly, the removal of the buffer tank eliminates a heat exchanger (a similar effect is noted, albeit in a different arrangement by Kelly and Hawkes [43]). Thirdly (and less significant), the deadband which is inherent in thermostatic radiator control results in a slight increase in the mean flow temperature at which heat is delivered (with mean weighted by heat delivered, not by time) for a given inside air temperature. The improvement potential associated with proportional control is an important finding that does not appear to be documented elsewhere and so it is suggested that further research should be carried out to confirm the viability of these savings. It should be noted that the improvements would not be so significant if modulating control of the ASHP units were not available (as has historically been the case).

Figure 5 compares the greenhouse gas emissions associated with the operation of ASHP A, ASHP B and the condensing gas boiler.

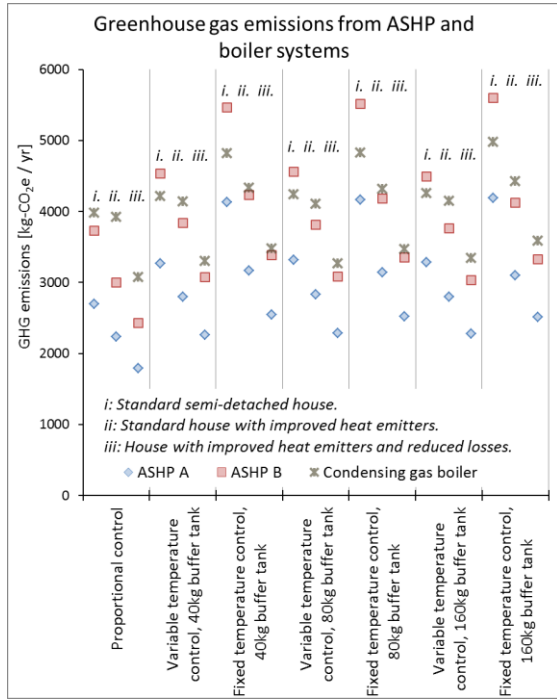


Fig. 5: GHG emissions

Changing the buffer tank size has minimal effect on the emissions associated with these systems. A very minor increase occurs with the gas boiler system due to increased losses but these are offset by the minimal performance gains in the case of the ASHPs.

Although it has been suggested that well-insulated houses are better suited to ASHP installations [8], this is not the case in some of the conditions considered here. Although reducing the heat loss from the building results in lower emissions, the reduction is not as great as that which occurs with gas boilers; the COP *decreases* (Figure 4). The effect is relatively minor and is caused by the redistribution of heating demands to times at which the outside air temperature is coolest. In the cases simulated here, there is limited potential to reduce the necessary flow temperature in the (already relatively cool) enhanced heat emitter system by reducing heat losses. However, in houses with higher temperature heat emitters it is likely that the conventional advice (i.e. that better insulation will improve performance) will still apply.

The impact of the control methodology (and of the inclusion of buffer tanks) on emissions is even greater than the effect on the COP. This is because the fixed-temperature system requires greater heat delivery (as well as decreasing the average COP of the system). Emissions increases of 45% to almost 55% are observed for the simulations using fixed-temperature control rather than proportional control. Although ASHP B can be operated with lower emissions than the gas boiler if it is proportionally controlled, its operation could result in greater emissions if it is used with variable- or fixed-temperature control in the standard building. Even with enhanced heat emitters, the use of ASHP B would result in only minimal emissions reductions if fixed-temperature control is used. The use of ASHP A results in lower emissions than the gas boiler in each case but the extent of the savings is highly dependent upon the conditions. Where single performance figures for emissions savings are reported, these should be treated with caution, especially if they are used to calculate marginal metrics such as the cost of carbon avoided.

3.2 SOFC-mCHP configuration

The unit efficiencies of the SOFC-mCHP unit tend to show small variations between the scenarios (Figure 6, relating to operation in the standard semi-detached house). In most cases the efficiencies are near to the optimum for the unit though there is some reduction in unit electrical efficiency when using full-range heat-led operation.

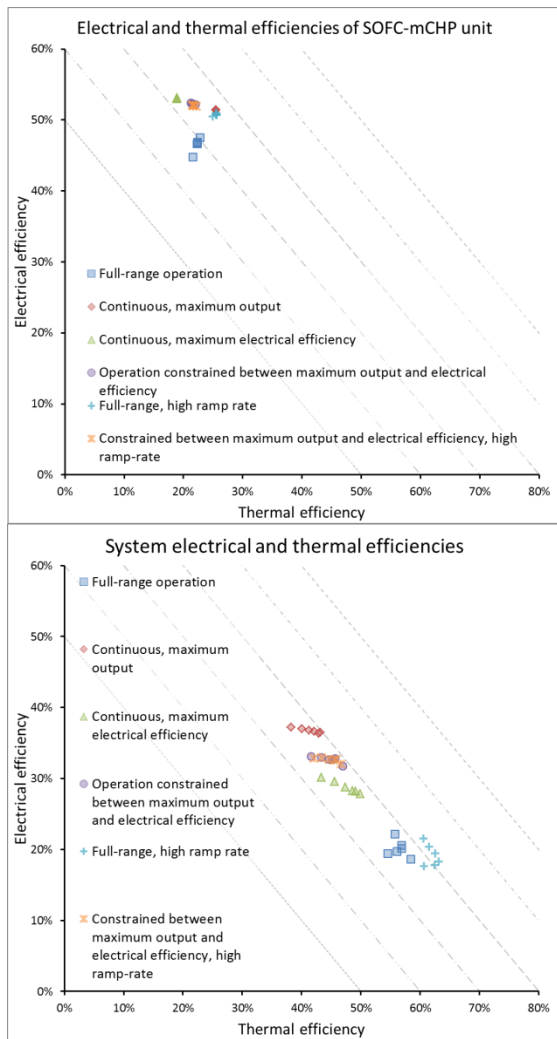


Fig. 6: SOFC efficiencies

However, if the system efficiency is considered, the characteristics change. These changes are primarily due to the inclusion of the auxiliary burner when the SOFC-mCHP unit cannot generate sufficient heat and so the magnitude of the changes would be reduced if a better insulated house was modelled. The highest system electrical efficiency is achieved by operating the unit continuously at full output rather than at its maximum unit electrical efficiency. Hypothetically removing the restrictions on the ramp rate of the units has minimal effect on the system electrical efficiency which is achieved.

The greenhouse gas emissions associated with the SOFC-mCHP systems operating in the standard semi-detached house are shown in Figure 7. Operating the unit continually at

maximum output (and dumping any heat which is surplus to requirements) results in the lowest net greenhouse gas emissions. In fact, the emissions in these scenarios are negative; the reduction in emissions from central electricity generation is greater than the direct emissions from the SOFC-mCHP unit and the auxiliary burner. Given this preferred mode of operation it seems that, in this application and under these circumstances, there is little to be gained (from an energetic or emissions perspective) in development effort aimed at improving the ramp-rate of the units.

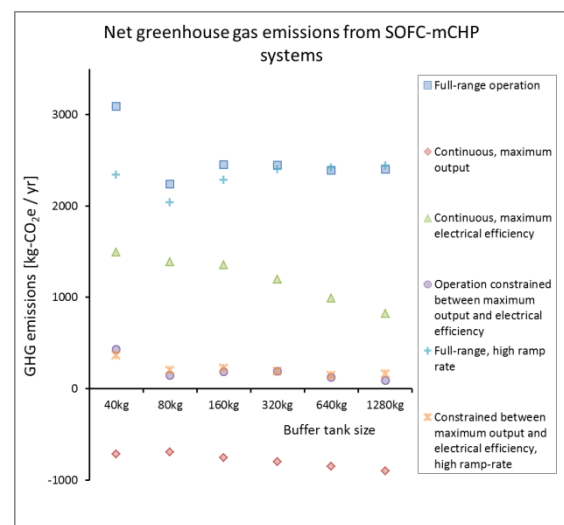


Fig. 7: GHG from SOFC systems

Figures 8 and 9 illustrate the performance of two variations on this scheme; operating the SOFC-mCHP continually but supplying heat to the different building specifications and with six buffer sizes.

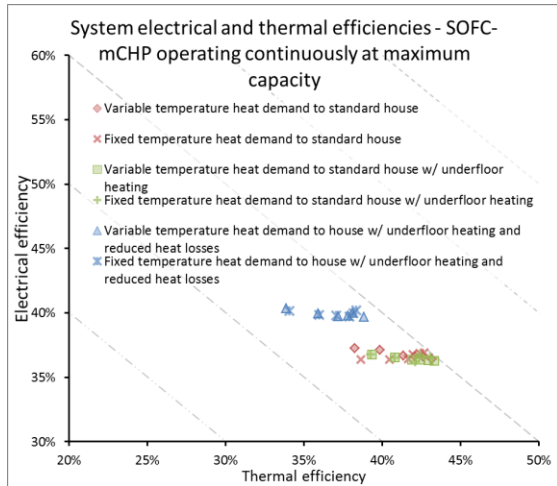


Fig. 8: SOFC system efficiencies

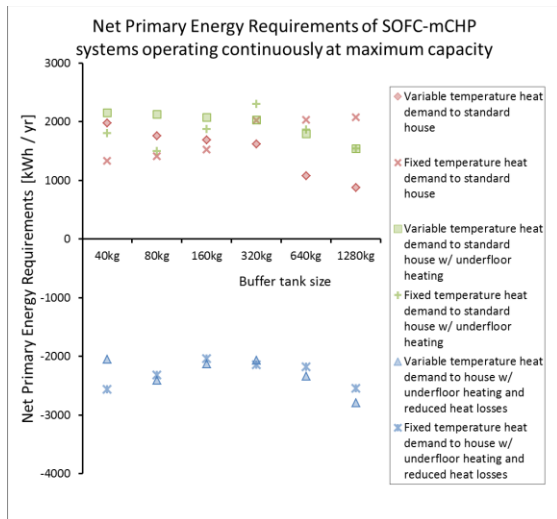


Fig. 9: PER of SOFC systems

The effect of the lower auxiliary heating demands in the buildings with reduced heat losses is to improve the system electrical efficiency in those scenarios as less fuel is used but the same amount of electricity is generated. The differences in system thermal efficiency (for each building type) therefore imply that different quantities of heat are delivered to each building under the different scenarios (and in some cases dumped). Although the increases in system electrical efficiencies in the reduced heat loss house are relatively small, they correspond to significantly lower (i.e. more negative) emissions. Although this specific result is sensitive to the grid emissions factor which is

used, the trend (lower emissions in the building with lower heat losses) is robust.

There are some minor differences between the use of fixed-temperature and variable-temperature control and a tendency for the mid-size (320 litre) buffer tanks to result in higher (i.e. less negative) emissions but the variations are relatively small. Increasing the size of the buffer tank results in less dumping of heat but also greater losses. The actual optimum size is likely to be sensitive to operational conditions that are not captured by this model.

3.3. Climate change

Figure 10 shows the effect of climate on the total heat (space heating and domestic hot water) supplied to the standard semi-detached house. The ASHP units and gas boiler are operated with proportional control while the SOFC-mCHP unit is operated continually at maximum output with a 1280kg buffer tank. The slightly higher quantity of heat supplied by the SOFC-mCHP systems in some scenarios indicates the extent to which they dump heat due to their continuous operation.

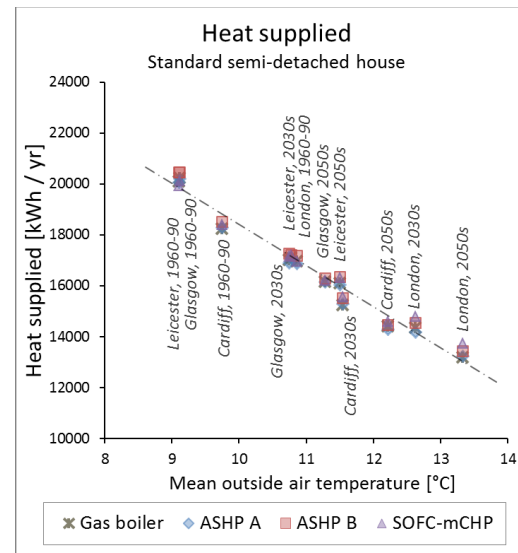


Fig. 10: Heat supplied, different climates

There is a general relationship in the modelled results between the heat demand and the mean outside air temperature although it should be

expected that hot water heating demand will become more significant as the average outside air temperature increases. There are slight discontinuities; the heat demands in Leicester and London (inland, standard deviation in temperatures of 6.2°C to 6.4°C and 6.0°C to 6.4°C) tend to be slightly higher than those in Cardiff and Glasgow (coastal, standard deviations of 5.6°C to 5.9°C and 5.7°C to 5.9°C) for a given average outside air temperature.

The effect of climate on the greenhouse gas emissions associated with satisfying these heat demands is shown in Figure 11. These use the same grid carbon factor so that the effect of climate is clearer but it should be noted that a lower emissions factor (hopefully corresponding to a future electrical grid) would decrease emissions from the ASHP systems and increase the net emissions from the SOFC-mCHP systems [23]. The emissions associated with the gas boiler decrease in proportion to the decrease in heat demand that occurs as average air temperatures increase. However, the average COPs of the ASHP units also increase (Figure 12), resulting in a greater decrease in emissions. It is possible that the increases in the average COPs relate more to decreases in flow temperatures than directly to increases in average outside air temperature.

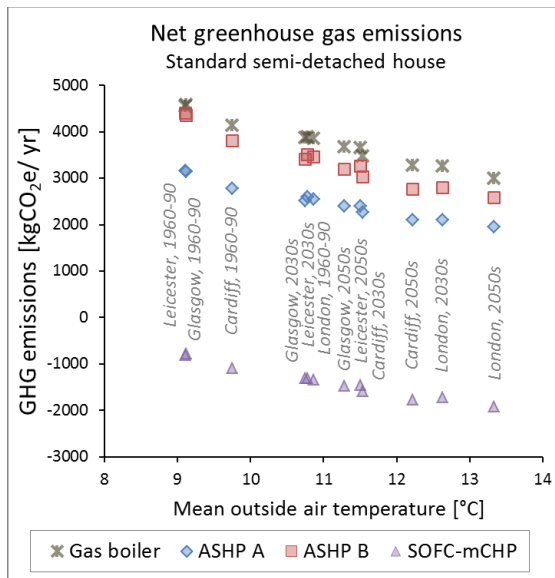


Fig. 11: GHG with different climates

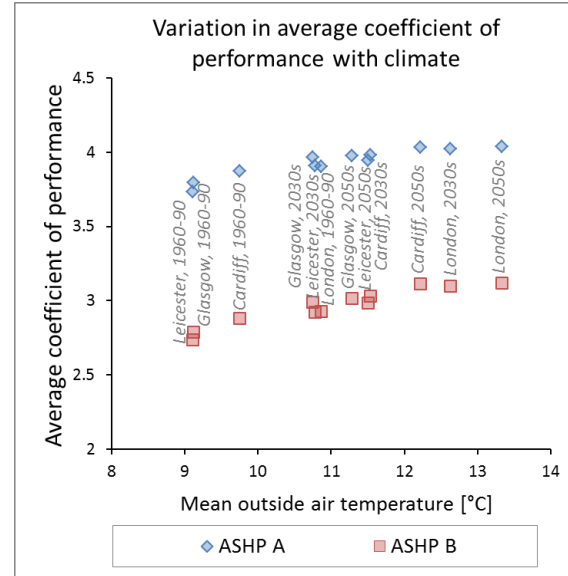


Fig. 12: Variation in COP with climate

In the case of the SOFC-mCHP systems, the lower heating demands tend to reduce the amount of heat which needs to be supplied by the auxiliary heating and so the reduction in emissions is also greater than that observed with the gas boiler systems.

These results relate to the performance of the systems in supplying heat. It is possible that increased cooling demands will mitigate savings.

4. Concluding remarks

A modelling approach has been taken to assess the relative performance of ASHP and SOFC-mCHP units under a range of conditions. The emphasis of this research has been to investigate the effect that the operational conditions have on the units and so it has been necessary to consider a wider range of permutations than in comparable studies. The effect of the control methodology employed with the units has been the primary focus of the study but other variables such as the climate and the specification of the buildings which heat is supplied to have been considered. These results can be used to inform the direction of more focussed simulation and research.

The performance improvements which might be achieved by using a proportional control methodology with the ASHPs are of particular significance; it is possible that they could reduce emissions by around a third. It is recommended that further research should consider this approach in more detail in order to verify its potential.

Single performance metrics for either ASHP or SOFC-mCHP units do not capture the range of energy performances which might be observed when the units are installed within complete heating systems. The effect of operational conditions should not be underestimated when comparing the relative merits of either technology to alternatives. Marginal metrics such as the cost of emissions avoided are even more sensitive to this variation if they are calculated from the difference in emissions between competing systems.

The effect of auxiliary heating units on overall system performance should not be underestimated and highlights the importance of the appropriate selection of boundary conditions when comparing heating options. For the SOFC-mCHP example considered in this study, the use of the auxiliary heater typically halved the electrical efficiency achieved by the system. A larger SOFC-mCHP unit for a given heat load would reduce this efficiency penalty but incur an increase in system costs.

In the context of displacing electricity generated by the UK electrical grid, operating the SOFC-mCHP units continuously at their maximum electrical generation capacity maximises the net emissions benefit that they achieve, despite the inevitable increase in heat dumping. If this objective (i.e. reducing net emissions) is adopted, then concerns regarding thermal cycling fatigue of the units and their ramp rates become less relevant. However, the insensitivity of this finding to the amount of heat which is dumped also implies that the main benefit of this technology is as efficient generation, regardless of whether it can be

employed in CHP schemes. Alternatively, it may be that another hybrid system including a SOFC-mCHP unit and an alternative system to cover peak heating demands would improve system performance.

The optimum configuration for ASHPs does not include buffer tanks. In contrast, the use of larger buffer tanks improves the performance of SOFC-mCHP units in most cases. All of configurations studied use separate domestic hot water tanks.

Assessments of the performance which these technologies might achieve in the future should take account of the climate change which might occur; that expected by 2050 in the UK is likely to reduce heating related emissions from ASHPs by around a quarter. The reduction in net energy requirements is likely to exceed the reduction in heat demand that a simple time-temperature difference model would suggest. Lower heat demands decrease the temperature at which ASHPs must deliver the heat. Lower heat demands also reduce the need for auxiliary heating in the case of the SOFC-mCHP systems, dramatically improving system electrical efficiencies. More varied air temperatures tend to result in higher heat demands for a given mean temperature but the overall trend remains.

Both ASHP and SOFC-mCHP units have the potential to contribute to reductions in energy use and the related emissions of CO₂. However, for this potential to be fully realised, the effects of operational conditions on the performance of both technologies should be understood and thoroughly researched.

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